

An experimental study on the development of a β -type Stirling engine for low and moderate temperature heat sources

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ABSTRACT

In this study, a β -type Stirling engine was designed and manufactured which works at relatively lower temperatures. To increase the heat transfer area, the inner surface of the displacer cylinder was augmented by means of growing spanwise slots. To perform a better approach to the theoretical Stirling cycle, the motion of displacer was governed by a lever. The engine block was used as pressurized working fluid reservoir. The escape of working fluid, through the end-pin bearing of crankshaft, was prevented by means of adapting an oil pool around the end-pin. Experimental results presented in this paper were obtained by testing the engine with air as working fluid. The hot end of the displacer cylinder was heated with a LPG flame and kept about 200 °C constant temperature throughout the testing period. The other end of the displacer cylinder was cooled with a water circulation having 27 °C temperature. Starting from ambient pressure, the engine was tested at several charge pressures up to 4.6 bars. Maximum power output was obtained at 2.8 bars charge pressure as 51.93 W at 453 rpm engine speed. The maximum torque was obtained as 1.17 Nm at 2.8 bars charge pressure. By comparing experimental work with theoretical work calculated by nodal analysis, the convective heat transfer coefficient at working fluid side of the displacer cylinder was predicted as 447 W/m² K for air. At maximum shaft power, the internal thermal efficiency of the engine was predicted as 15%.

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1. Introduction

For the current situation, fossil fuels meet most of the world's energy demand, but because of the rapid depletion of fossil fuels and environmental considerations, the interest in alternative energy sources and effective energy conversion systems has significantly increased [1–3]. The energy conversion from solar radiation or any heat energy to mechanical energy can be performed via Stirling engines with high thermal efficiency. The Stirling engine is a mechanical device working with a closed cycle. In the theoretical cycle of the Stirling engine, the working fluid is compressed at constant temperature, heated at constant volume, expanded at constant temperature and cooled at constant volume [4–6]. Since the Stirling engine is externally heated and the heat addition is continuous, many energy sources, such as solar radiation, combustible materials, radioisotope energy, all kind of fuels so on, can be used. Stirling engines have less pollutant emissions in comparison with internal combustion engines [4,7–9].

The Stirling engine was first patented in 1816 by Robert Stirling. Stirling engines built in 19th century were huge in volume and small in power. Modern Stirling engines are environmentally

harmless and thermally more efficient however, still there are some problems to solve. In the last few decades, Stirling engines operating under low temperature difference, especially solar powered low temperature difference Stirling engines, gained so much importance. These engines can be run with very small differences in temperature between the hot and cold sides of the displacer cylinder and are able to use low grade, cheap and waste heat sources including solar energy [9–12].

Since 1816, many studies on Stirling engines have been conducted. The era of modern Stirling engine development was started in 1937 by the Philips Company of The Netherlands. Philips developed a number of Stirling engines of various sizes up to 336 kW [12,13].

In 1983, Kolin [11] demonstrated the first low temperature difference Stirling engine. Senft [11,14] developed the Ringbom engine using the ideas introduced by Kolin. Iwamoto et al. [15] compared the performance of a high temperature difference Stirling engine with a low temperature difference Stirling engine. They concluded that at the same working conditions the thermal efficiency of the low temperature difference Stirling engines will not reach that of high temperature difference Stirling engines.

Kongragool and Wongwises [16] manufactured and tested twin power pistons and four power pistons, gamma-configuration low temperature difference Stirling engines. Engines were tested with

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Nomenclature

s_c	length of cold volume (m)	T_i	nodal values of working fluid temperature (K)
s_h	length of hot volume (m)	U_c	length from cylinder top to the fixing pin center (m)
$\frac{H_p}{2}$	distance between piston top and piston pin (m)	B_d	angle made by displacer rod with cylinder axis (rad)
H_d	displacer length (m)	β_p	angle made by piston rod with cylinder axis (rad)
l_d	length of displacer rod (m)	γ	angle made by slotted arm of lever with cylinder axis (rad)
l_m	distance of fixing pin and crank pin (m)	$(\frac{\pi}{2} - \varphi)$	Conjunction angle of lever arms (rad)
l_R	length of displacer rod (m)		
l_p	length of piston rod (m)		
S_L	length of lever arm connected to displacer rod (m)		

various heat inputs using a domestic gas burner. Power of engines measured were far-off any industrial application.

Cinar and Karabulut [9] designed and manufactured a gamma type Stirling engine. The engine was tested with air and helium. Maximum output power was obtained as 128.3 W. Karabulut et al. [17] investigated the thermodynamic performance characteristics of a novel mechanical arrangement of concentric Stirling engine using nodal analysis. In an experimental study, carried out by Cinar et al. [18], a β -type prototype Stirling engine was manufactured and tests were conducted at atmospheric pressure. The engine provided maximum 5.98 W brake power at 208 rpm.

A Stirling radioisotope generator having 110 W power output is currently being developed by the Department of Energy, Lockheed Martin and the NASA Glenn Research Center to use in space investigations. This generator promises higher efficiency, specific power and reduced mass compared to alternatives [19].

This study aims to develop a Stirling engine for low and moderate temperature energy sources having 200–500 °C heating range. The power output of the engine was designed as 500 W. As working fluid ambient air and helium was considered. The speed of the engine was designed as 1200 rpm. The prime objective of the engine to be developed is solar power generation at domestic scale. The engine should be eligible to couple with a parabolic collector and to reflect solar rays directly onto the hot end of displacer cylinder.

In most of solar energy applications of Stirling engine, the solar rays are concentrated by a parabolic dish and focused on a heat pipe. The solar energy received by the heat pipe is converted to heat and conveyed to the heater or hot end of the Stirling engine. The heat pipe is used to reduce the energy losses by thermal radiation and reflection as well as satisfying the uniform heating. A Stirling engine working at 200–250 °C hot end temperature is not exposed to high thermal stresses. The energy loss by thermal radiation is also not too much. Therefore, in a domestic scale solar energy conversion system the use of a Stirling engine working at 200–250 °C hot end temperatures may exclude the heat pipe and simplify the system as well as reducing the cost of construction. The engine being developed is also considered for water pumping in greenhouses.

2. Mechanical arrangement

In Fig. 1 and Table 1, the mechanical arrangement of the manufactured β -type Stirling engine and its specifications are shown. The cylinder of the engine consists of two sections connected to each other end-to-end. The first part functions as piston liner and made of oil hardening steel. The second part functions as displacer cylinder and made of ASTM steel. The displacer and the displacer cylinder wall were used as a regenerator. Two different displacer cylinders were manufactured and tested. One of them has a smooth inner surface and the other has augmented inner surface with rectangular slots having 2 mm width and 3 mm depth. The in-

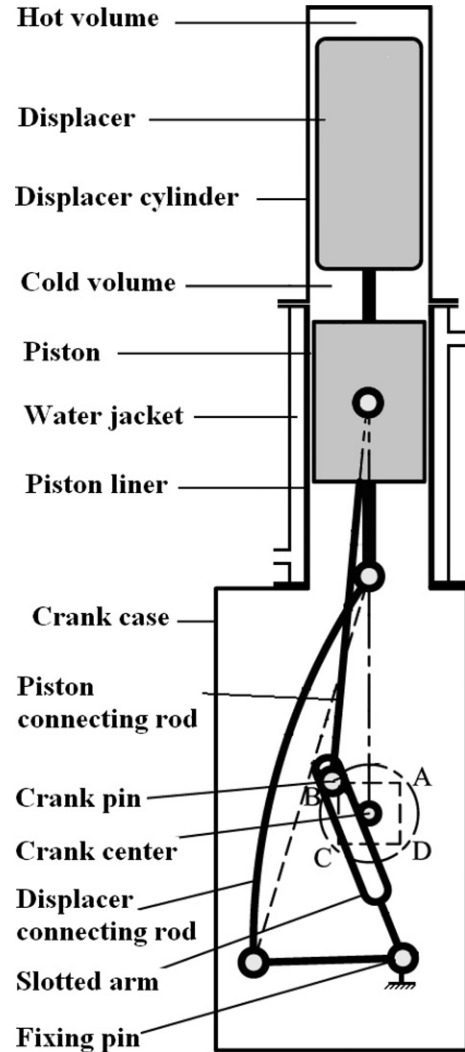


Fig. 1. Schematic illustration of the test engine.

ner surface of the piston liner was finished by honing. The external surface of the piston liner is cooled by water circulation. The piston was made of aluminum alloy because of its light weight and machining simplicity. The surface of the piston was machined at super-finish quality. Appropriate value of clearance between piston and piston liner was determined experimentally.

As shown in Fig. 1, the crankshaft has only one pin for the connection of piston rod and lever arm. The motion of the displacer is governed by a lever. The lever-controlled mechanism and the assembly of piston and displacer are shown in Fig. 2. The lever consists of two arms with 70° conjunction angle. One of them holds a

Table 1
Technical specification of the test engine

Parameters	Specification	
Engine type		β
Power piston	Bore $\times$ stroke (mm)	70 $\times$ 60
	Swept volume (cc)	230
Displacer	Bore $\times$ stroke (mm)	69 $\times$ 79
	Swept volume (cc)	295
Working fluid		Air
Cooling system		Water cooled
Compression ratio		1.65
Total heat transfer area of the displacer cylinder (cm ²)		1705
Maximum engine power		51.93 W (at 453 rpm)



Fig. 2. The lever-controlled mechanism, piston and displacer assembly.

slot bearing as the other holds a circular bearing. At the corner of lever, a third bearing exists. The lever is mounted to the body of engine through the bearing at the corner by means of a pin. The slot bearing at one arm of the lever is connected to the crank pin. The circular bearing at the other arm of lever is connected to the displacer rod. While the crank pin turns around the crank center, it drives the lever fort and back around the pin connecting the lever to the body. The other arm connected to the displacer rod moves the displacer up and down. Both ends of crankshaft were bedded with ball bearings. Escape of working fluid through the crankshaft bed was prevented by means of adapting an oil pool around the crank shaft end-pin. As long as the charge pressure was below 5 bars no air leak was observed. The block of the engine was manufactured of two parts and coupled by screws. Between two parts of the engine block a plastic seal was set. To lubricate working parts of the engine some oil was filled into the engine block and its circulation was satisfied by throwing via the lever arm.

3. Kinematic relations

Thermodynamic analysis was conducted by using the nodal program presented by Karabulut et al. [17]. The working space of the engine was divided into 50 nodal volumes. To calculate nodal values of hot and cold volumes, kinematic relations

$$S_c = l_d \cos \beta_d - S_L \sin(\gamma - \varphi) + l_R - l_m \cos \gamma - l_p \cos \beta_p - \frac{H_p}{2}, \quad (1)$$

$$S_h = U_c - l_d \cos \beta_d - S_L \sin(\gamma - \varphi) - l_R - H_d \quad (2)$$

were used. In these equations β_p is the angle between piston rod and cylinder axis and defined as

$$\beta_p = \text{Arcsin}\left(\frac{R_{cr}}{l_p} \sin \theta\right). \quad (3)$$

β_d , γ and l_m are; the angle between displacer rod and cylinder axis, the angle between slotted arm of lever and cylinder axis and the distance between lever's fixing pin and crank pin, respectively. Mathematical definitions of them depend on the location of the lever's fixing pin. The point providing the maximum work per cycle was chosen as the location of fixing pin and determined via isothermal analysis. With respect to the location of fixing pin; β_d , γ and l_m were defined as

$$\gamma = \text{Arctan}\left(\frac{0.7071 + \sin \theta}{2.5 + \cos \theta}\right), \quad (4)$$

$$\beta_d = \text{Arcsin}\left(\frac{S_L}{l_d} \cos(\gamma \varphi) - \frac{R_{cr}}{l_d} 0.7071\right), \quad (5)$$

$$l_m = \frac{R_{cr}(0.7071 + \sin \theta)}{\sin \gamma}. \quad (6)$$

The conjunction angle of lever arms was defined as $(\frac{\pi}{2} - \varphi)$ and the optimum value of φ was determined as 0.35 rad. Nodal volumes within the regenerator have constant values and take place around the displacer.

4. Experimental apparatus and testing procedure

A prony type dynamometer with accuracy of 0.003 Nm was used for loading the engine. The speed of the engine was measured by a digital tachometer, ENDA ETS1410, with 1 rpm accuracy. Temperatures were measured with a non-contact infrared thermometer, DT-8859, with ± 2 °C accuracy. Heat was supplied by a LPG burner. The charge pressure was measured with a bourdon tube pressure gauge with 0.1 bar accuracy and 0–10 bars measurement range. A pressure regulating valve was used for the regulation of the charge pressure. The schematic view of the experimental apparatus is shown in Fig. 3.

The charge pressure was applied to the block of the engine and its value was read from the pressure gauge. By means of increasing and decreasing the external load, the speed of the engine was stabilized at any desired value and then reading of the load; speed and temperature were made simultaneously.

5. Results and discussion

The variation of cold volume, hot volume and total volume with the crank angle is shown in Fig. 4. In the engine the minimum value of cold volume appears about 50° of crank angle and, before and after 50° it performs significant variations. This interval of crank angle corresponds to expansion period of working fluid. The interval of crankshaft angle from 135° to 225° corresponds to cooling process at constant volume. In this process the mechanism presents a better performance by means of minimizing the hot volume and maximizing the cold volume without changing the total volume.

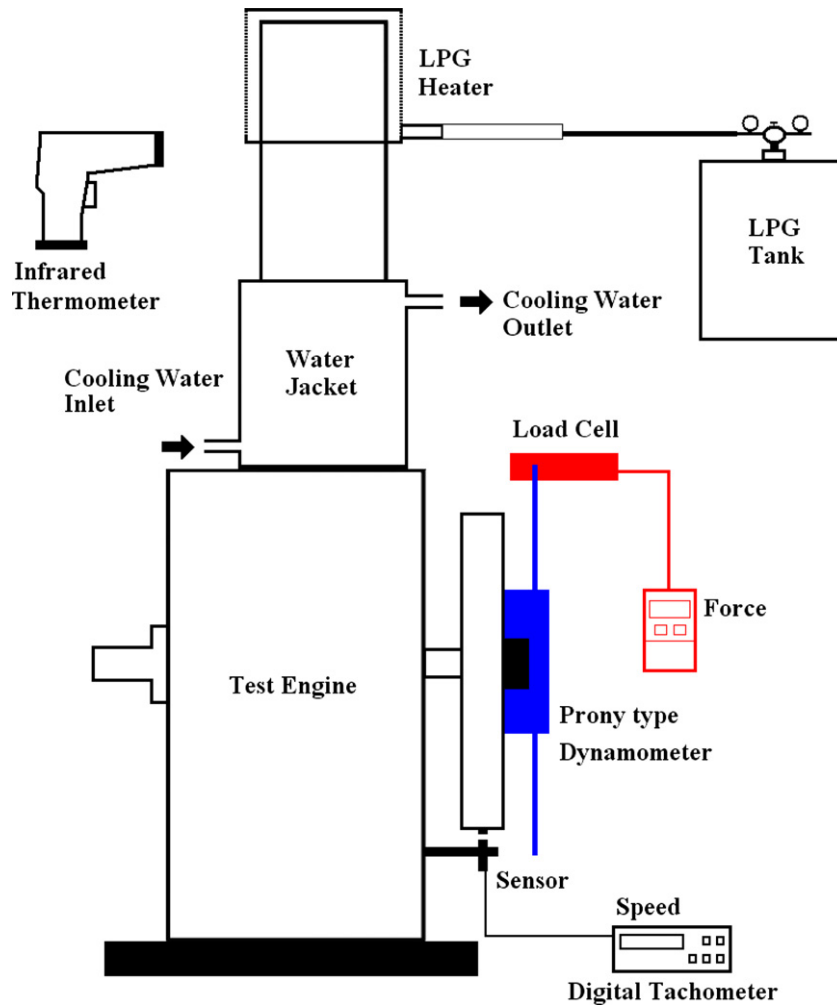


Fig. 3. Schematic illustration of the test equipment.

At initial testing conducted with ambient pressure and displacer cylinder having smooth inner surface, the engine started to run at a 93 °C hot-end temperature. Cooling water temperature was measured as 27 °C. At the other tests conducted with different charge pressures, running temperature of the engine varied up to 125 °C. When the heating was ceased, the engine continues to run until the hot-end temperature drops to 75 °C.

For the displacer cylinder having augmented inner surface, the variations of power output and brake torque with engine speed

are illustrated in Fig. 5 and 6, respectively. Data used in Figs. 5 and 6 were obtained about 200 °C hot-end temperature and at different values of charge pressure. Up to a certain level of speed, the power increases with speed and then declines. Decrease of the power output after a certain speed is estimated due to inadequate heat transfer caused by limited heating and cooling time. Flow and mechanical frictions may also have effects on decreasing of power. As shown in Fig. 5, the power output obtained at 3.5 and 4.6 bars are lower than at 2.8 bars. At 200 °C hot-end temperature, the optimum charge pressure is estimated as 2.8 bars. Maximum power output obtained at 2.8 bars charge pressure is 51.93 W and appears at 453 rpm engine speed. As seen in Fig. 6, the brake torque has also a maximum value at a certain value of speed. At higher and lower values of speed, the reasons causing the power output to decrease causes also the brake torque to decrease. The maximum values of power and torque correspond almost to the same speed.

In the nodal program developed by Karabulut et al. [17], by using 2.8 bars charge pressure, 200 °C hot-end temperature, 27 °C cold-end temperature and 453 rpm engine speed, the convective heat transfer coefficient at working fluid side of cylinder was determined as 447 W/m² K by trial and error corresponding to 51.93 W shaft power or 6.88 J shaft work per cycle. The *p*–*V* diagram of the engine obtained for these conditions was illustrated in Fig. 7. In the same figure, the *p*–*V* diagram obtained by isothermal analysis, which corresponds to an infinite heat transfer coefficient, was also illustrated. In order to check whether the heat transfer coefficient of 447 W/m² K is valid for the other cases or not, the nodal pro-

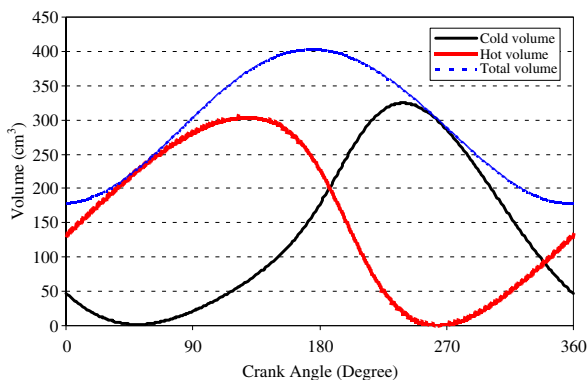


Fig. 4. Variation of cold volume, hot volume and total volume with crank angle.

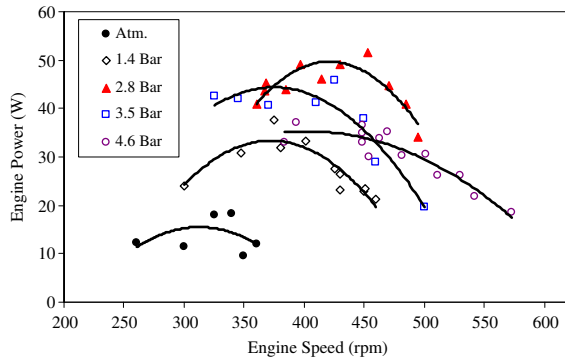


Fig. 5. Variation of brake power with engine speed.

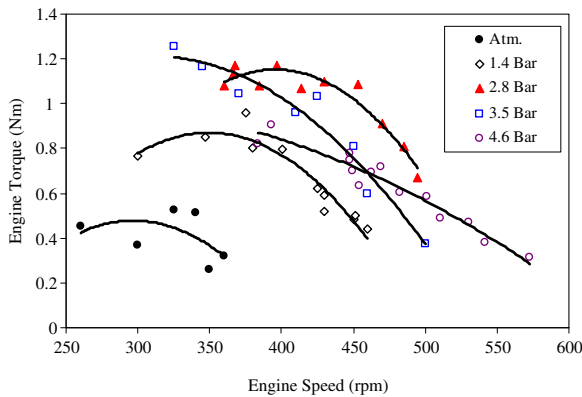


Fig. 6. Variation of engine torque with engine speed.

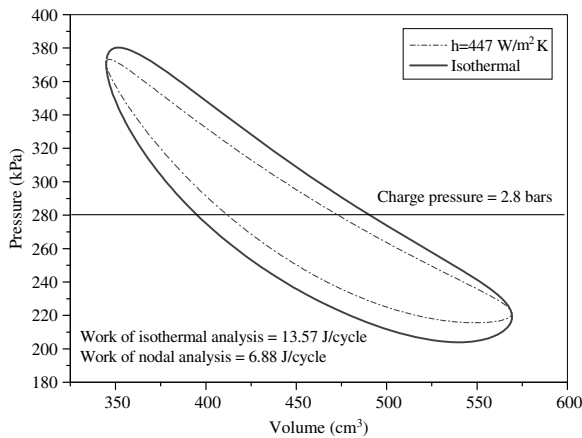
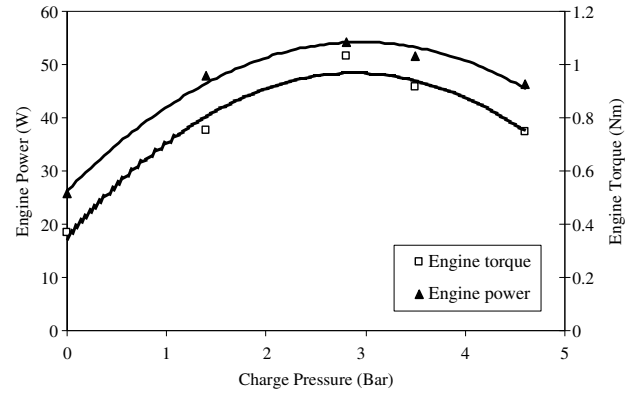
Fig. 7. p - V diagrams obtained with isothermal and nodal analysis.

Fig. 8. Variation of brake power and engine torque with charge pressure.

By means of equating Nusselt numbers of air and helium to each other, the heat transfer coefficient of helium was calculated as $2392 \text{ W/m}^2 \text{ K}$. If we consider that Nusselt number increases as flow velocity increases, the heat transfer coefficient of $2392 \text{ W/m}^2 \text{ K}$ is a safe value to predict the power of the engine with helium as working fluid. By means of introducing 2.8 bars charge pressure, 500°C hot-end temperature, 27°C cold-end temperature, 1200 rpm engine speed and $2392 \text{ W/m}^2 \text{ K}$ heat transfer coefficient to the nodal program, the power of the engine was predicted as 493 W. In addition this, if we consider the increase of charge pressure with hot-end temperature, the engine should provide a higher power than the above predicted.

Fig. 8 illustrates the variation of the power output and brake torque with charge pressure ranging from 0 to 4.6 bars. As the charge pressure increases, the output power and brake torque increase as well, and reaches to a maximum. Further increase of charge pressure over the optimum value causes output power and brake torque to decrease. It was also noted that, increasing the charge pressure resulted in increase of vibration.

Figs. 9 and 10 show the performance comparison of smooth and slotted displacer cylinders in terms of engine power and brake torque. Both cylinders were tested at 200°C hot-end temperatures and several values of charge pressure. The slotted cylinder having 214% larger inner surface than smooth cylinder provides about 50% higher power which is lower than our expectations. The reason limiting the power and torque is guessed as inadequacy of hot-end temperature and decrease of compression ratio. To get larger power outputs, the hot-end temperature should be increased or the augmentation of inner surface of the displacer cylinder should be made without permeating the compression ratio to decrease at a significant rate.

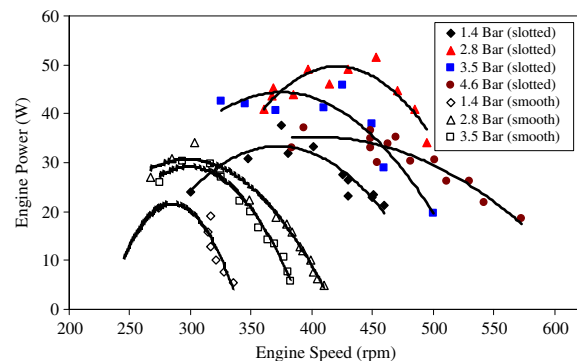


Fig. 9. Comparison of smooth and slotted cylinders in terms of power.

gram was run with 325 rpm engine speed, 1 bar charge pressure and $447 \text{ W/m}^2 \text{ K}$ heat transfer coefficient. As the result 4.068 J work was obtained. For the same conditions, the experimental value of work is 3.32 J which can be deduced from Fig. 5. The difference between these experimental and theoretical works may be caused by mechanical frictions which consumes a certain amount of work produced by working fluid. Obviously while the engine runs at a low power, the ratio of mechanical losses to the produced work by working fluid becomes larger. Therefore, the convective heat transfer coefficient determined as $447 \text{ W/m}^2 \text{ K}$ seems to be valid for different situations caused by engine speed, charge pressure and compression ratio. Obviously, variation of the slot geometry and the gap between displacer and cylinder will cause the convective heat transfer coefficient to vary.

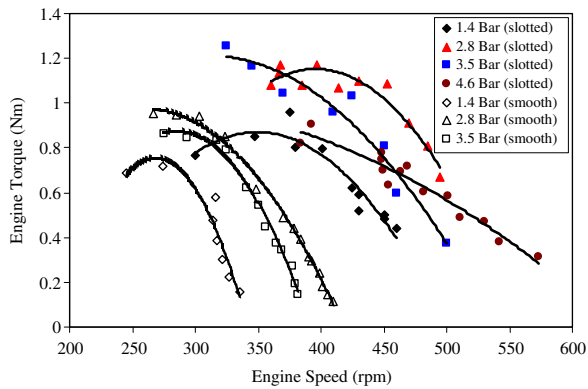


Fig. 10. Comparison of smooth and slotted cylinders in terms of torque.

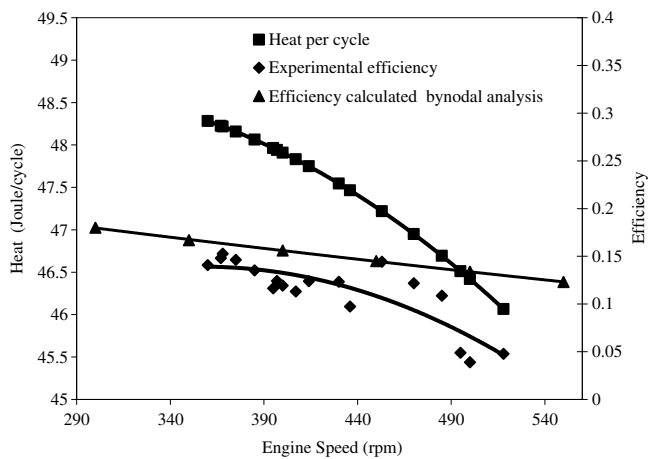


Fig. 11. Thermal efficiency and heat given to the working fluid.

If the variation of internal thermal efficiency of the engine is examined by nodal analysis using the convective heat transfer coefficient of $447 \text{ W/m}^2 \text{ K}$, the charge pressure of 2.8 bars, the hot end temperature of 200°C , the cold end temperature of 27°C and specific values given in Table 1, a linear variation with engine speed is obtained as seen in Fig. 11. The thermal efficiency varies from 12% to 18% as the engine speed varies from 550 rpm to 300 rpm. Determination of the heat transferred to the working fluid per cycle was not easy to measure. Therefore the heat was predicted by nodal analysis by means of introducing experimental conditions to the program and results were illustrated in Fig. 11. By using the heats predicted by nodal analysis and experimental works, the thermal efficiency of the engine was determined and compared to the theoretical efficiency in Fig. 11. While the experimental efficiency approaches to the theoretical efficiency at low speeds of the engine, it presents larger diversity at high speeds. The difference may be caused by several reasons but mainly inadequate heat transfer to and from working fluid.

While the charge pressure decreases, thermal efficiency of the engine increases. For instance, corresponding to 3.32 J shaft work obtained at ambient pressure testing, the efficiency of the engine was calculated as 20.6%. At lower charge pressures however, the ratio of mechanical losses to the work generated by working fluid will increase and the break thermal efficiency will decrease.

6. Conclusions

By means of a lever controlled displacer driving mechanism a better approximation to theoretical Stirling cycle was achieved. By using a LPG burner as heat source, the engine was tested with air up to 4.6 bars charge pressure. At the ambient pressure testing, the engine started to run at 93°C hot-end and 27°C cold-end temperatures. On the starting temperature the charge pressure made a slight effect. Maximum power output was obtained as 51.93 W at 453 rpm engine speed and 2.8 bars charge pressure. The internal thermal efficiency corresponding to maximum power was determined as 15%. When the heat transfer area of the engine was increased by augmenting the displacer cylinder inner surface, a 50% increase in output power was obtained. Via an oil pool adapted to the crankshaft's end-pin bearing, the leak of working fluid was perfectly avoided up to 5 bars charge pressure. For 500°C hot-end temperature, 27°C cold-end temperature, 2.8 bars helium charge pressure and 1200 rpm engine speed, the power of the engine was predicted as 493 W via the nodal program.

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